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# VENTILATION STUDY OF PROPOSED AMMUNITION PLANT AT LAKE CITY ARSENAL.

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FOREWARD

The research discussed in this report was accomplished as part of the Safety Engineering Graduate Program conducted jointly by the USAMC Intern Training Center and Texas A&M University. The ideas, concepts and results presented are those of the author and do not necessarily reflect approval or acceptance by the Department of the Army.

This report has been reviewed and is approved for release. For further imformation on this project contact Dr. George D. C. Chiang, Chief of Safety Engineering, Red River Army Depot, Texarkana, Texas 75501.

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procedure is applied to an Ammunition loading plant at Lake City Arsenal where explosive dust is generated. Results show that a proposed 75% recirculation system does provide a significant savings over the original 100% make-up design. A system using 90% recirculation, however, was found to be the optimum design. Restrictions and recommendations are also discussed.

# ABSTRACT


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A procedure has been presented that can be used to compare ventilation costs of systems utilizing different rates of recirculation. Systems are designed so that they all provide equivalent internal conditions. Air exchange rates that provide an acceptable contaminant concentration are determined for each recirculation rate. Refrigeration and air washer requirements along with appropriate costs are then determined for each design. This procedure is applied to an Ammunition loading plant at Lake City Arsenal where explosive dust is generated. Results show that a proposed 75% recirculation system does provide a significant savings over the original 100% make-up design. A system using 90% recirculation, however, was found to be the optimum design. Restrictions and recommendations are also discussed.

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# CHAPTER I

## INTRODUCTION

The problem of airborne dust in certain working environments still exists, although numerous advances in control methods have been made in recent years. In the past, workers were exposed to different dusts without knowledge of the toxic effects they had on the respiratory or biological system. Today, however, both acute and chronic effects are known for many substances as a result of extensive testing. The Industrial Hygienist is able to provide protection for the worker by knowing the hazardous effects of these dusts and utilizing the set standards for the maximum allowable dust concentrations.

Many dusts exhibit a hazard due to their explosive nature in addition to their toxic effects. This fact magnifies the problem of providing a safe environment for the worker. Unlike the situation where the dust contributes only to a toxic effect and Threshold Limit Values (TLV's) can be exceeded providing the time weighted average is within the limits, the maximum allowable explosive dust concentration must not be exceeded. Extreme care in controlling dust concentrations must be taken when dealing with explosive dust.

Airborne explosive dust is a problem in ammunition manufacturing facilities. Here explosive propellant dust is generated in areas of shell assembly. Control methods, including isolation of the processes and local ventilation in the form of vacuum pickups, are used to minimize the amount of dust being emitted to the workroom air. While these methods minimize dust emission, additional methods must be used to control the small portion that is released. General ventilation is often used to control the dust emission if this rate is small.

Selection of ventilation system design is a major problem when general ventilation is relied upon for this purpose. The problem is further complicated if the ventilation system must also relieve heat stress by conditioning return air. Industrial Hygienists often face the problem of selecting a ventilation system design that will maintain desirable conditions while keeping the cost at an acceptable level. A procedure to be used for this purpose is presented and applied to an ammunition manufacturing plant.

The building to be considered in this ventilation study is a proposed facility at Lake City Army Ammunition Plant, Independence, Missouri. It will house the production equipment necessary for the manufacture of small caliber ammunition. Because the production assembly to be used is similar to those in existing facilities, problems with explosive dust concentrations in the worker environment are known. Explosive dust, as used here, refers to the dust generated from explosive material as opposed to fine dust with explosive properties. It will be used as such throughout this study.

Independence, Missouri is geographically located in an area where both high temperatures and humidity are encountered in the summer months. The ventilation system must thus provide incoming air conditions that reduce worker heat stress in addition to reducing explosive dust concentrations.

According to John Spencer, Safety Director at Lake City Army Ammunition Plant, Independence, Missouri (27)* two major air conditioning units are being considered for this ammunition plant. The recommended unit is a 100% make-up air system while the alternative is a 25% make-up air unit utilizing 75% filtration and recirculation. As the name indicates, the 100% make-up air unit draws its entire supply of ventilation air from an external source such as the outside air. As this air is brought into the working area, an equal amount of inside air is exhausted along with the suspended explosive dust. The large amount of make-up air required with the 100% make-up air system along with the large temperature gradient between outside air and conditioned spaces combine to place a large heat load on the conditioning system.

The 25% make-up air unit, on the other hand, will take only 25% of its ventilation air from an external source.

*Numbers in parenthesis pertain to references at end of report.

the remaining 75% ventilation air will be taken from the working area. This air will be filtered before being mixed with the outside air to form the ventilation air. It is evident that the air conditioning unit required with this design is much smaller than the 100% make-up air unit due to the reduced heat load encountered. It is the lower initial and operating costs of the smaller unit that make it desirable for use in this facility.

Although the 25% unit is desired for use in this facility, it is not known if it will be able to provide the required work environment. Systems utilizing recirculation, return some unfiltered dust to the workroom area. Humidity may also be returned if the cooling unit design is inadequate. These factors must be considered when analyzing ventilation systems.

The objectives of this project are twofold. The first objective is to determine whether or not the 25% make-up air unit, which was decided upon for use in this facility, can provide an acceptable environment. The second objective is to determine whether or not this is the optimum system as regards percent make-up air. If it is not, the optimum percent make-up air system will be determined. It should be noted that the air conditioning system used in this facility must meet certain design criteria: It must be able to provide necessary cooling to compensate for encountered heat loads, humidity must be maintained at a level conducive to ammunition manufacture, and explosive dust concentrations must be maintained at an acceptable level.

The initial cost of the air conditioning apparatus along with the operating performance cost will be investigated for various systems utilizing different make-up air percentages. These figures will be used to determine if the 25% make-up air unit is a feasible alternative to the 100% unit, and to determine the optimum system.

# CHAPTER II

## PROBLEM DEFINITION AND GENERAL PROCEDURE

Ammunition production is the result of many unrelated procedures occuring in an orderly fashion. Perhaps this can be made more clear by first looking at Figure 2-1 to review the main parts of a shell or round as they are commonly referred. It can be seen from this drawing that a shell consists of a case, a primer cup, the primer, the propellant and the bullet. Each of these parts is manufactured in a separate division of the production facility.

The significance of having separate locations for the various manufacturing processes is to prevent hazardous conditions in one area from readily combining with those from another resulting in a more hazardous situation.

### Manufacturing Process

In order to understand the problems associated with the ventilation of such a facility, the individual manufacturing processes along with the assembly of the manufactured components must be considered. The case, which is the major component of the assembled cartridge is produced from a small cup shaped piece of brass which is 70% copper and 30% zinc. (20) These cup shaped pieces undergo numerous processes as they proceed by conveyor through the metal forming apparatus before taking their final shape as shown in Figure 2-2a.. Bullets and primer cups are made in a similar fashion. These

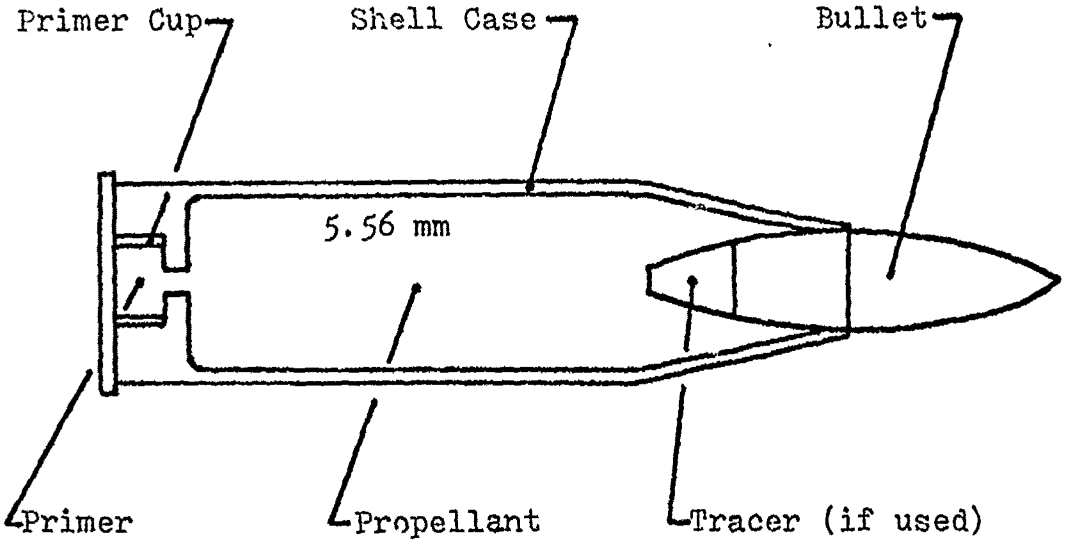



Figure 2-1

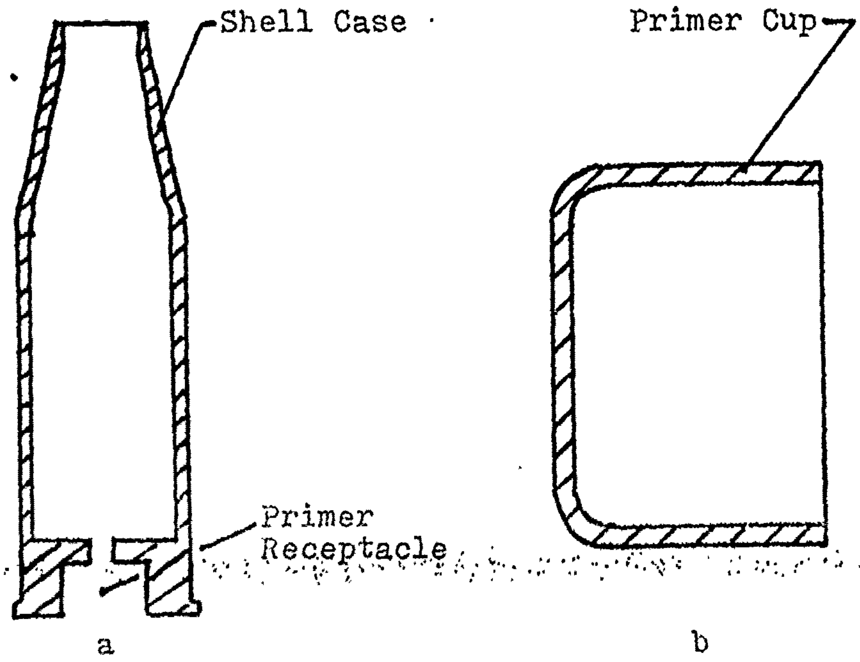


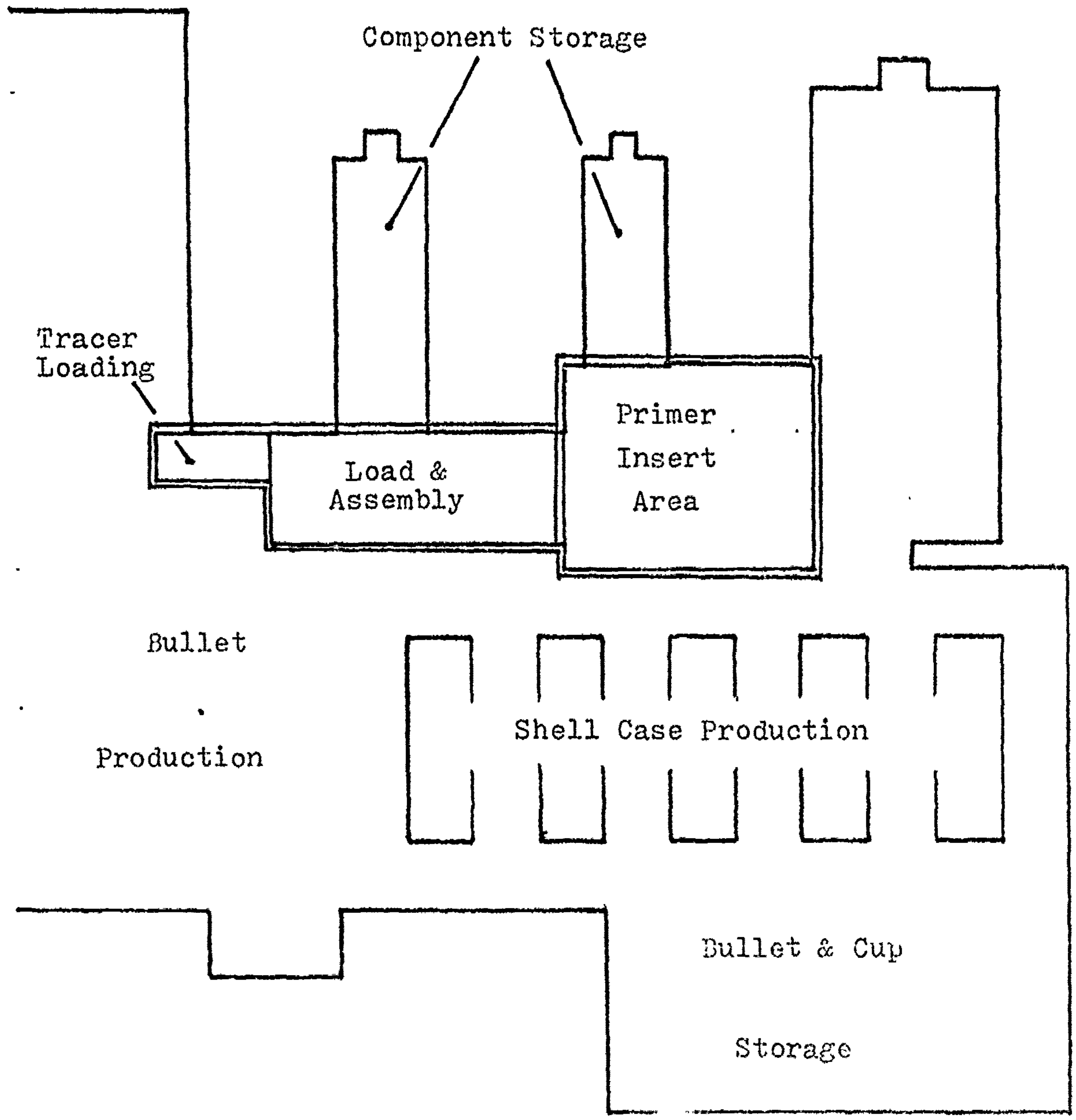

Figure 2-2

processes take place in the areas shown in Figure 2-3. After the above pieces are produced, they are conveyed to the primer insert and load and assembly areas where the shell is completed.

The first step in the assembly of a shell once the necessary components are manufactured, is to prime the case. Primer is a substance that is very sensitive to pressure but is not very powerful once exploded. (20) It is used solely to initiate the explosion of the propellant. This substance is placed in the primer cup as shown in Figure 2-4 followed by a piece of paper and the anvil. Once this primer is assembled, it passes along to the primer insert area where it is inserted in the receptacle of the shell case as the case passes by on a conveyor in an inverted manner. The shell case is then crimped around the primer to hold it in place. It is this process of primer insertion and crimping that sometimes inadvertantly detonates the primer thus causing dust to be admitted into the surrounding air. In fact, the dust from primer detonation is the sole source of contamination in the primer insert area.

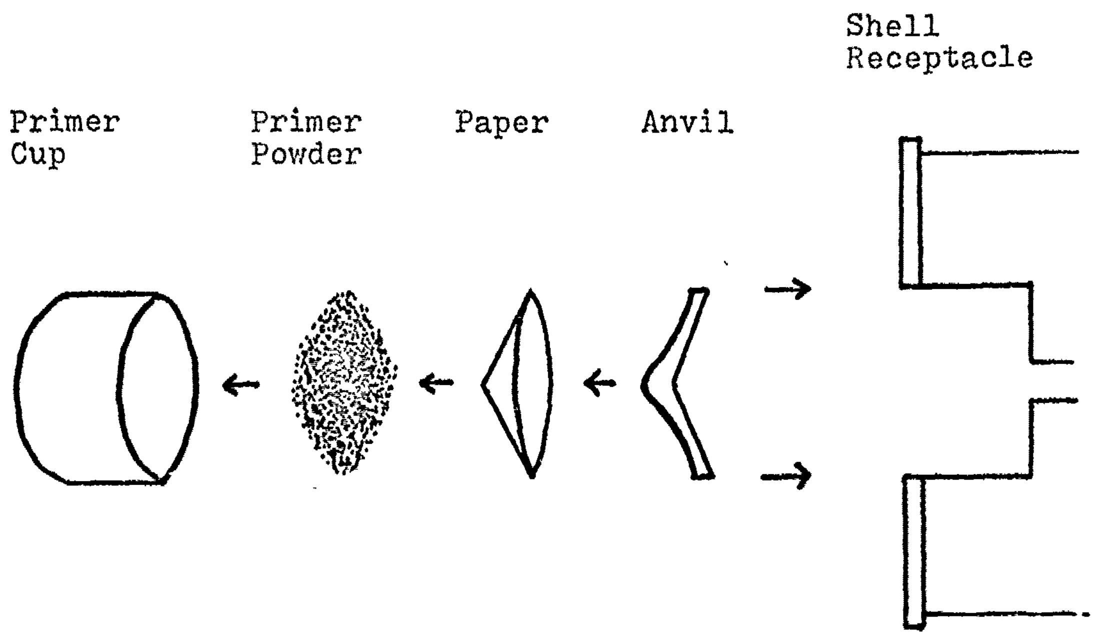
After the primed cases reach the load and assembly area they are turned mouth side up and conveyed beneath a gravity feed hopper. Here a measured amount of propellant is dropped into each case. (27) A mechanical device is used to check the amount of propellant in each shell before it is allowed to pass down the assembly line. Once loaded, the shell is

Plant Layout




5.56 mm cartridge manufacture

Figure 2-3




The above diagram shows the processes involved in priming the shell case. Primer is inserted into the primer cup as shown followed by a piece of paper and the anvil. This entire apparatus is then inserted into the case receptacle. The base of the shell case is then crimped around the primer cup.

Diagram Taken From Ordinance Production Methods by Charles O. Herb

Figure 2-4

enclosed by having the bullet inserted and crimped in place further down the line. Shellac and markings are applied to the shell and the process is complete. The gravity feed hopper mentioned above is the source of explosive dust emission in the load and assembly area. Due to its open design, loose dust from ball and tracer propellants are dispersed by air currents created by ventilation, infiltration, or manufacturing processes. It is the dust from this operation along with the priming operation that is to be controlled by ventilation in this study.

Problem Area

Priming and loading areas are the only locations in the plant where explosive dust is generated and therefore, should be isolated from other work areas. The main reason for confining the explosive dust is to prevent this dust from combining with hazards in other locations, producing more serious hazards. An example of this is the situation where explosive dust is allowed to come into contact with sparks, static electricity, or hot surfaces generated by the case or bullet forming machinery. This occurrence greatly increases the chance of explosion and therefore, should be eliminated.

Isolation of the priming and loading areas is achieved by partitioning these areas from the rest and providing them with a separate ventilation system. Separate ventilation is required to eliminate the dispersion of dust through the ventilation network as would result if a common system were used. Priming and loading areas also require separation from each other. This is to prevent propellant dust from entering the priming area and being ignited by the primer detonation. For these reasons, priming and loading areas will be considered individually throughout this study.

<u>Procedure</u>

In conducting such a ventilation study, the first step should be to thoroughly analyze the problem area. The sources of contaminant generation along with the rates of emission should be determined. In some cases, where the operation is totally new, estimates of these values may be required. These estimates should, however, be conservatively set. An acceptable dust concentration must also be established. This is the concentration of dust in the work area that ventilation must provide under the most severe dust emissions. Both explosive and toxic characteristics of the contaminant at hand should be considered in setting this value. An acceptable dust concentration should be chosen that is lower than either of these respective TLV's.

The procedure to be followed once these values have been determined is to determine the required air flow rates for ventilation designs utilizing various rates of recirculation that will provide the same acceptable dust concentration. The desirable characteristic of such a method is that, all systems under consideration will provide the same work environment and can be compared on a system size or cost basis. It is apparent that the dust concentration selected above is the basic design parameter, as required air flow rates are dependent upon this value. Care should be taken in selecting this concentration to ensure a ventilation system that is optimally designed.

Before air flow rates can be determined for systems using recirculation, filter parameters must be known. Appropriate filters can be selected once the type, emission rate and acceptable concentration of the contaminant have been determined. The minimum expected efficiency of the selected filter can then be used to calculate air flow rates.

After the air flow rates that will provide an acceptable dust concentration have been derived, the problem is to estimate the conditioning equipment needed to provide acceptable thermal conditions. This problem should be attacked by first deciding upon desirable work area temperatures and humidity levels. Representative values of extreme external environmental conditions to be encountered must also be established. When these values are decided upon, heat loads due to solar transmission, normal transmissions. internal sources and infiltration can be calculated.

Ventilation air must balance these heat loads if desirable workroom conditions are to be maintained. The exact temperature and humidity levels of this air can be determined from heat loads, air exchange rates, and desired workroom conditions already established.

Conditions of the ventilation air prior to entering the cooling unit can also be established by using workroom and external air conditions along with the rate of recirculation. Cooling requirements can then be calculated utilizing the entering and leaving air conditions at the cooling unit along with the required mass air flow rate. This procedure should be applied to each system design so that cooling and air cleaning requirements are established for all systems under consideration. Costs can then be assigned to each design so that a comparison, to determine the optimal system, can be made.

# CHAPTER III

## DISCUSSION OF DESIGN CONDITIONS

In order to calculate the cooling and humidification requirements for the various ventilation systems under study, the design conditions mentioned in Chapter 2 must be determined. Representative numerical values of these conditions will be discussed in this chapter. Dust emission rates, filter selection, area and volume calculations, inside and outside environment conditions, and heat gain calculations will be included.

### Dust Emission Rate

Estimates of explosive dust emission were to be obtained from Twin Cities Army Ammunition Plant because similar equipment is used there. According to Harold Wright, (34) Safety Director at this facility, however, no appreciable explosive dust concentration could be measured in loading areas. Because most of the propellant grain sizes are quite large, 99% above 250 $\mu$, they will fall to the loading table and be removed by vacuum pick ups. Due to crushing and abrading of larger grains, a small portion of the dust will be in the 1-10 $\mu$ range. This dust may cause excessive dust concentrations if proper ventilation is not provided. Although an estimate of dust emission was not obtained, it was found that this emission is extremely small.

Black & Veatch, Consulting Engineers (8) designed the original 100% make-up air ventilation system. In their study, they used .03mg/$M^3$ as the maximum acceptable design concentration. This is the concentration of dust in the air that will be provided by the ventilation system during periods when dust emission is at a maximum. By using this concentration and Black & Veatch's air exchange rate, the maximum acceptable emission rate can be determined using a method to be shown later. The emission rate, found by this procedure, is quite a bit larger than would be expected from Wright's report. This difference is the safety margin provided by the ventilation system.

When dust emission is lower than this calculated maximum value, as it apparently is, dust concentration will be much lower than the design concentration. This is a desirable condition, as the EPA (16) states that the dust concentration should be kept as low as possible. Estimates of dust emission rates based on Black & Veatch's data will be used in this study so that conditions provided by the systems under study will be the same as the original 100% design. This will provide the same safety margin and will enable comparison of the systems under study with the original 100% make-up design.

To determine this maximum rate of dust emission, we make use of a fundamental material balance equation. This balance, as stated by Mutchler, (22) is (rate of accumulation = rate of generation - rate of removal). Stated in equation form for our system, we get

$$VdC = G'dt - QCdt \qquad 3\text{-}1$$

Where:
- $V$ = Volume of room
- $C$ = Concentration of dust in room
- $Q$ = Ventilation rate
- $G$ = Generation rate
- $K$ = Design distribution constant
- $G'$ = $GK$ = effective generation corrected for incomplete mixing of generated dust & room air.

By using this equation and the fact that concentration of dust in the air is constant, hence, $dC=0$ we see that

$$G' = QC \qquad 3\text{-}2$$

This equation can be used only if certain criteria are met. First of all this relation holds if the generation of contaminant is continuous and at a constant rate. Secondly, the emitted contaminant must disperse readily throughout the air like a vapor or gas.

Dust emission from propellant loading areas occurrs each time a shell is loaded and hence, is a discreet process. Because the number of shells loaded per minute is quite large and each loading emits the same small amount of dust, the processes are assumed to be continuous thus meeting condition one above. Particles, unlike gases and vapors, do not diffuse in the classical sense, but rely upon air movements to distribute them. Drinker and Hatch (13) state that particles in the range of 1-10$\mu$ are distributed readily by air movement. Because the dust under consideration is in the range of 1-10$\mu$ (9) and the assumption that the ventilation duct design used provides adequate air distribution, condition two above is assumed to have been met. For these reasons equation 3-2 will be used to calculate the dust generation rate for the loading, tracer charge, and sample bullet assembly areas. In the primer insert area, dust is emitted in discreet amounts. This dust is generated when accidental detonation occurrs, which from past experience, occurrs an average of 2 times every hour. (27) Each time detonation occurrs, 1.5 grains (27) of dust particles are produced: this is equivalent to a dust emission rate of .1946 g/hr.

Although the primer substance in its raw form is extremely sensitive and explosive, the dust due to detonation is not. For this reason the design dust concentration may be exceeded providing the average is within the accepted limits. Assuming 2 detonations per hour, each emitting 1.5 grains of dust, the average emission rate is .003243 g/min. This value is also found to be much less than the estimate provided by Black & Veatch's data, as shown below. The larger value will be used to provide a safety margin.

A. Propellant Loading Area

Using the value obtained from Black & Veatch (8) and equation 3-2 we get the maximum allowable effective dust generation rate G.

$$Q = 30{,}000 \text{ cfm}$$

$$C = .03 \text{ mg/M}^3$$

$$G' = QC = .02548 \text{ g/min.}$$

B. Tracer Charging Area

$$Q = 3{,}736 \text{ cfm}$$

$$C = .03 \text{ mg/M}^3$$

$$G' = QC = .00317 \text{ g/min.}$$

C. Sample Bullet Assembly Area

$$Q = 7{,}263$$

$$C = .03 \text{ mg/M}^3$$

$$G' = QC = .00617 \text{ g/min.}$$

D. Primer Insertion Area

$$Q = 32{,}100 \text{ cfm}$$

$$C = .03 \text{ mg/M}^3$$

$$G' = QC = .02727 \text{ g/min.}$$

The emission rates determined in this section will be used in Chapter 4 to determine the required ventilation rates for various make-up air systems.

Air Cleaner Selection

The ventilation systems under study use recirculation for some portion of the ventilation air. This recirculated air will require cleaning to remove explosive dust. Fabric and membrane filters, gravitational settling chambers, cyclones, electrostatic precipitators and wet cleaners will be considered for use.

The cleaner must be able to remove airborne explosive dust, in the 1-10$\mu$ range, from a large volume of air. Volume flow rate will be in the range of 20,000 - 50,000 cfm for each cleaner while the dust concentration will be .03 mg/$M^3$. These criteria will be used to select an appropriate cleaner.

The electrostatic precipitator is immediately rejected from consideration. The high potential energy used in this system may cause ignition of airborne or collected dust.

While fabric and membrane filters do have a high collecting efficiency, certain aspects of their design make them objectionable. Filters of this type, exhibiting high efficiency, have a large resistance to air flow. In order to reduce the total resistance, a large surface area is required with a low air velocity. Drinker and Hatch (13) state that a velocity of 3 fpm is optimal for filters of this type. For a 30,000 cfm air rate, a filter with an area of 10,000 $ft^2$ would be required. Fabric filters also have the inherent hazard of allowing dangerous levels of explosive dust to be collected by the filtering medium if not maintained properly. High concentrations of dust in the filter not only increase the chance of explosion but also increase the potential damage if explosion occurrs.

Self cleaning fabric filters are available. They function by forcing air through the filter in a reverse direction thus dislodging particles trapped in the filter medium. This type of system has the unwanted characteristic of allowing dislodged particles to collect in the filter housing. (23) For this application, they are not recommended.

Gravitational settling chambers are not feasible for dust of this size and type. The dimensions of such a unit would be extremely large. According to Drinker and Hatch (13) an air velocity of 60 fpm is considered optimal for such a system. For an air flow of 30,000 cfm, a cross sectional area of 500 $ft^2$ would be required for this velocity. To remove particles as small as 1 $\mu$, the ratio of height to length of such a unit would be 1-7400 (13). It is apparent that a filter of this type would not be feasible.

Cyclones, that are able to handle large volumes of air, are not very efficient at removing particles below 10 $\mu$. Small cyclones on the other hand, are able to remove small particles but cannot handle large volume air flows. For these reasons, dry cyclone systems will not be considered as a means of cleaning air in this facility.

Wet collectors, being the last type of system to be discussed, are considered the best type of air cleaning device for this facility. They adequately remove particles in the desired range while eliminating the hazard of dust build up in the cleaner by utilizing a continuous cleaning action. Although the cost of these units is quite high relative to the cleaners mentioned earlier, the benefits out-weigh the cost disadvantage.

In the original 100% make-up air design, an air cleaner was to be used to filter the exhaust air before emitting it to the atmosphere. Although designed mainly for exhaust air, it will be used for the recirculated air in our design. This system makes use of a water spray as shown in Figure 3-1 (8). As air passes through the stream, dust is entrained by the water droplets. This combined mixture is then passed through a cyclone where the dust laden water droplets are separated from the air by centrifugal force. Clean air passes out of the system while the collected water and dust drains at the bottom of the cyclone. From here the water is passed through a separate bag-type powder sump where the dust is removed from the water in a continuous fashion.

Systems of this type, according to Jones (21), can have efficiencies of 97% for particles down to 1$\mu$. For particles smaller than 1$\mu$ the collecting efficiency decreases so that .5$\mu$ particles are collected with 90% efficiency. Because only a small portion of the particles are below 1$\mu$, we could assume the collecting efficiency to be 97% under ideal conditions. Jones (21), however, states that systems of this sort can have efficiencies varying by as much as 4% when operating at less than design capacity. For this reason the cleaning system will be assumed to have a minimum efficiency of 93%.

Area & Volume Calculations

Before we can determine the cooling load requirements for the areas under consideration, the floor area and room volume must be determined. These values are shown in Figures 3-2 to 3-5.

Air Washer

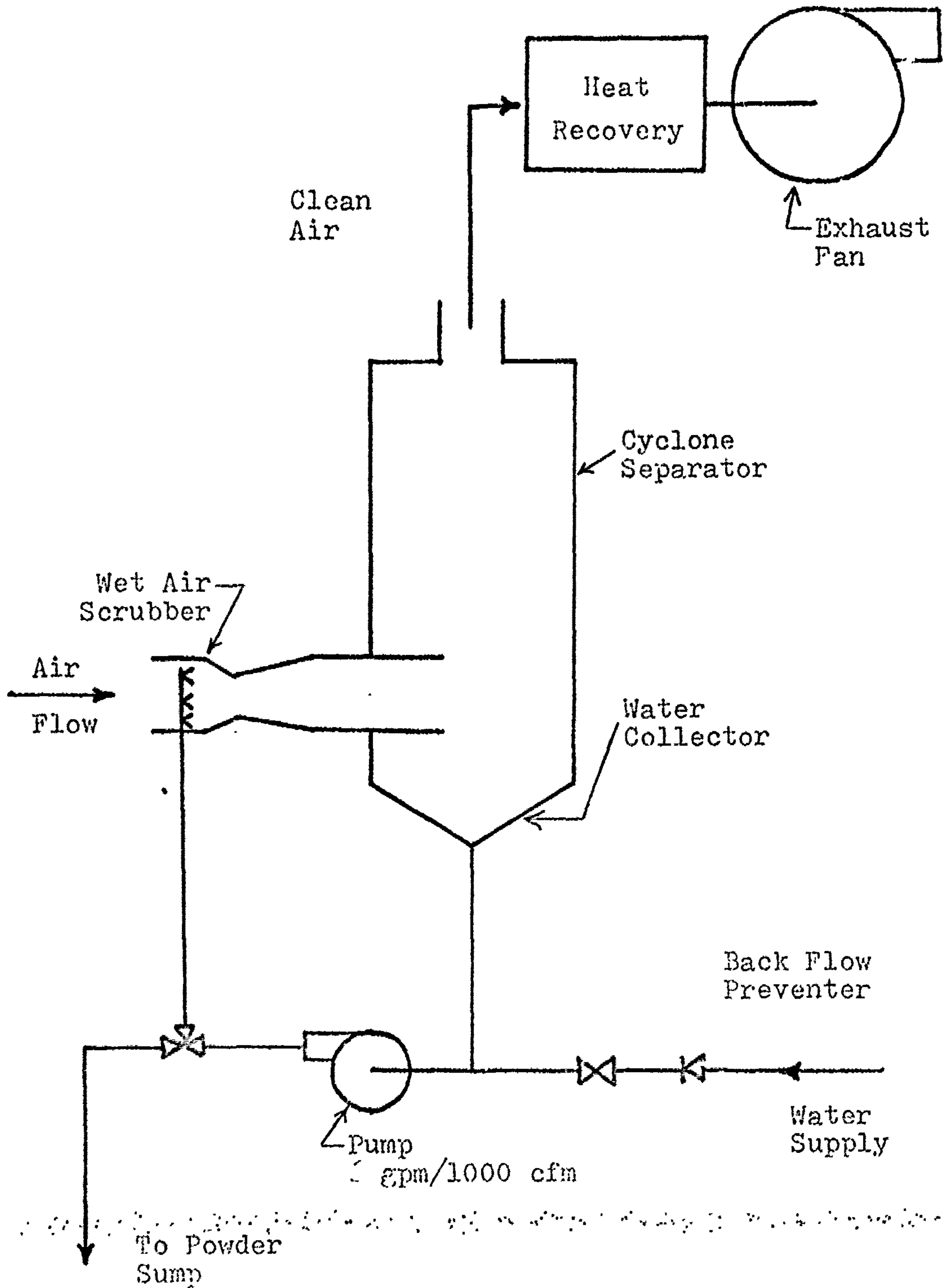



Figure 3-1

Primer Insert

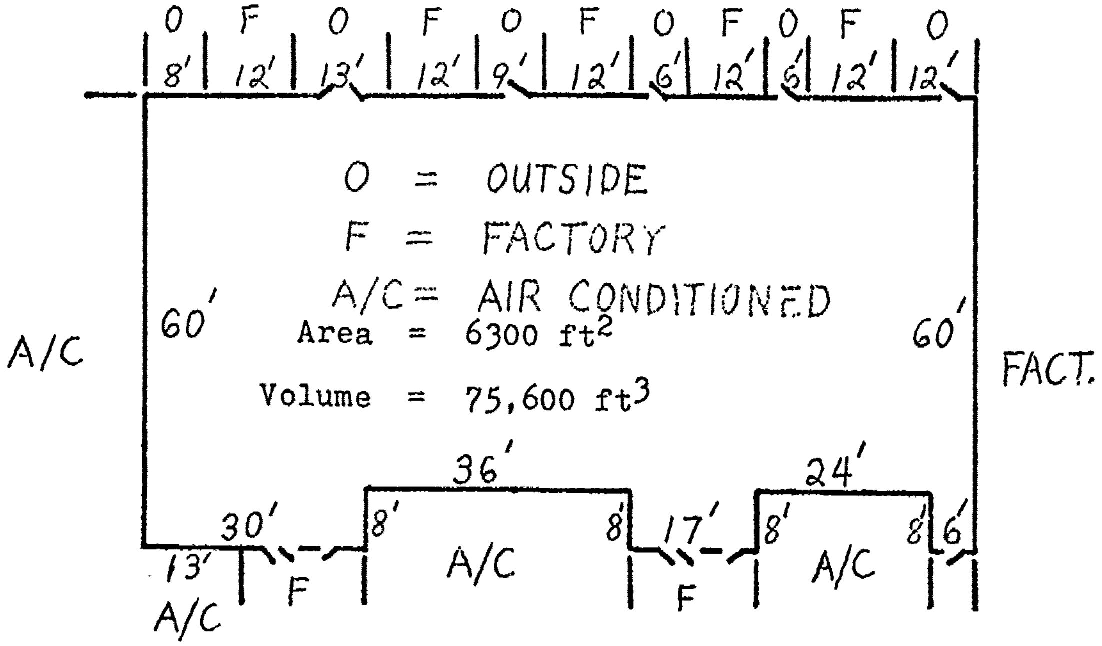



Figure 3-2

Load & Assembly

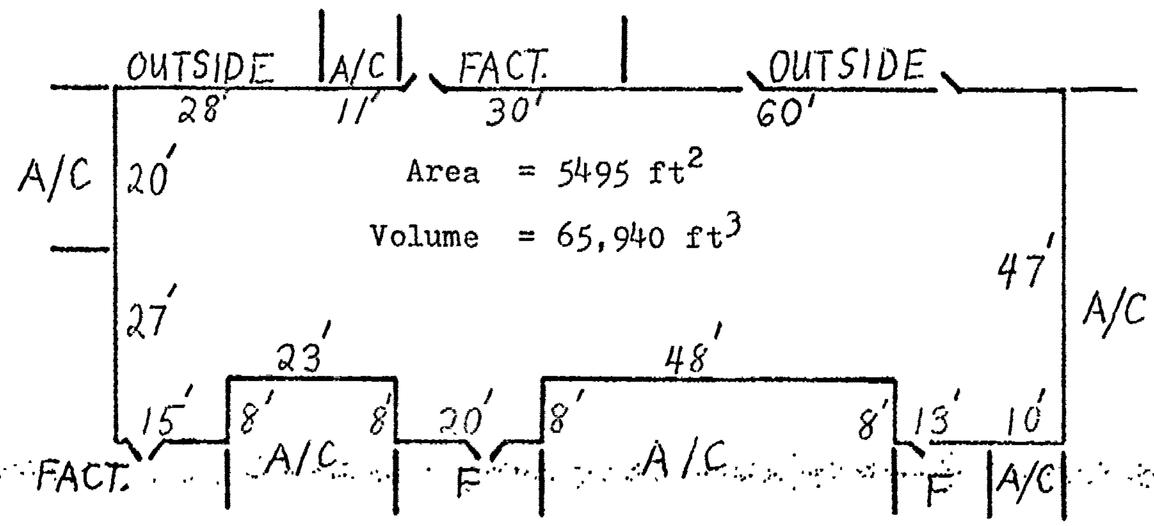



Figure 3-3

Tracer Charge

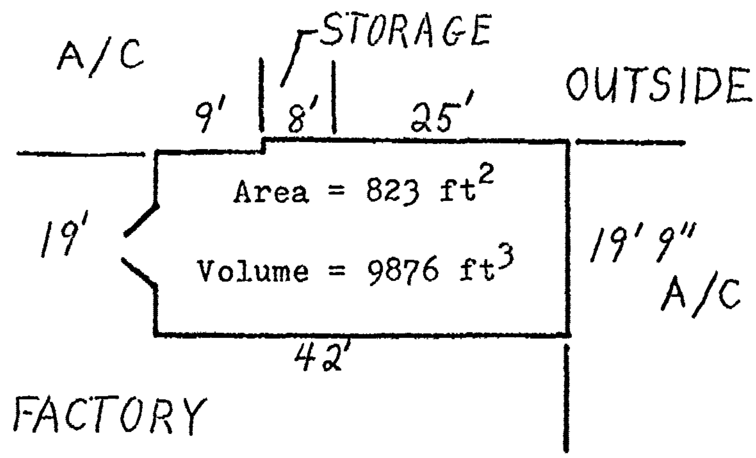



Figure 3-4

Sample Bullet Assembly

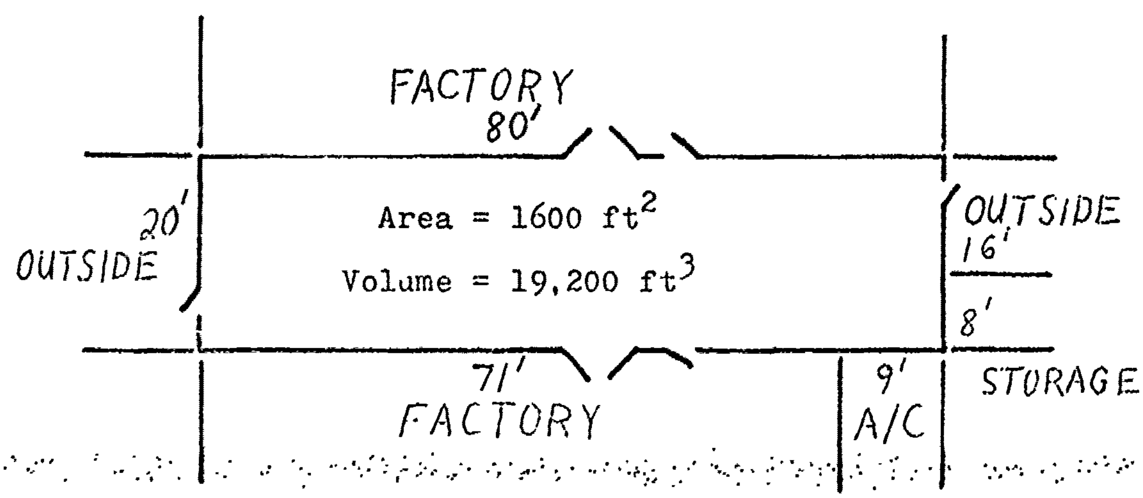



Figure 3-5

## Environment Conditions

An estimate of the required cooling can be made only after work area heat gains have been calculated. These heat gains are directly related to the inside and outside environment conditions. Appropriate values of dry-bulb temperature and relative humidity must thus, be chosen for these spaces. The importance of selecting representative values cannot be stressed enough. Incorrect selection could lead to an uncomfortable work area at one extreme or a vastly overdesigned system at the other.

### Inside Conditions

Selection of conditions desirable for this facility depend upon a number of factors namely; the type of manufacturing processes being conducted, and the level of physical exertion by workers. Propellant handling area require close control of humidity. Excessive humidity degrades propellant performance while low humidity enhances static electricity generation, which is a potential source of ignition. A compromised value of 50% relative humidity has been selected as the design value.

Using the fact that comfort can be provided by infinite combinations of dry-bulb temperature, relative humidity and air velocity, we will find an acceptable dry-bulb temperature corresponding to a relative humidity of 50% and an air velocity of 50 fpm. (31) A velocity of 50 fpm was chosen because ventilation rates and room volume dictated so.

Using Fanger's comfort charts (5) for a man doing moderate work a temperature of 73° is found. A dry-bulb temperature of 73° and a relative humidity of 50% were chosen as design conditions.

Outside Conditions

The selection of appropriate outside design conditions is an important step in the estimation of required air conditioning equipment. Unlike the situation where inside conditions can be arbitrarily chosen, outside conditions are governed by nature. For this reason, past weather data will be used in determining acceptable values.

According to Clifford Strock (29), it is wrong to select design dry-bulb and wet-bulb temperatures separately. This is due to the fact that maximum wet bulb and dry-bulb temperatures never occur simultaneously. Separate selection of these design conditions could result in an overdesigned cooling unit be as much as 50% (1).

A wet-bulb temperature of 78° has been chosen as the design condition for this facility. This value will be exceeded 1% of the time in summer months but, is considered acceptable. From past weather data, a maximum dry-bulb temperature of 101° is found to correspond to the design wet-bulb temperature and will be used as the design dry-bulb temperature. This value of 101° is found to be exceeded 2½% of the time in summer months (1). When this value of 101° is exceeded, however, the corresponding W.B. temperature is less than 78$^{o}$ so the internal heat gain should not be excessive.

Heat Gain Calculations

Heat gain calculations can be made after appropriate environment conditions have been selected. The encountered heat gains will be from four major sources, each of which will be discussed separately. These sources include solar and normal transmission, internal loads, and infiltration.

Solar and Normal Transmission

In most facilities, the heat gain due to solar radiation represents a significant portion of the total heat gain. Solar radiation, however, plays a rather small role in the heat gain of the areas under study. The reason for this is that most of the wall and roof surfaces of the spaces under study are not exposed to the external environment. Instead, they are exposed to either air conditioned or non air conditioned areas of the facility as indicated by Figures 3-2 through 3-5.

With surfaces where no solar transmission takes place, the only heat gain will be normal transmission. This heat transmission is due to the differing air temperatures and hence, surface temperatures on opposite sides of the wall or roof. Steady state conditions will be assumed for these areas in the estimation of heat gain. This assumption can be made when temperatures on both sides of the wall or roof remain constant. Heat gain can then be calculated by using the relation below. (31)

$$q = U \times \Delta T \times A \qquad 3\text{-}3$$

Where: $q$ = Heat gain

$U$ = Material transmission coefficient (BTU per hr. - $ft^2$ - $^oF$)

$\Delta T$ = Temperature difference of surfaces on opposite sides of the wall or roof (To -Ti)

$\Delta T = (101^o - 73^o) = 28^o$ in this study

$A$ = Total area of the wall or surface with like conditions

Although most wall and roof surfaces fall into the above category, some surfaces are exposed to the outside air. For these surfaces, steady state conditions cannot be assumed because outside temperature does not remain constant. Threlkeld (31), recommends the use of an effective temperature difference ($\Delta Te$) in equation 3-3 for such surfaces. The effective temperature difference makes corrections for solar transmission gains, heat storage of the material and the temperature changes in the outside air.

### Internal Loads

The majority of the total heat gain in the spaces under study is due to internal sources. These sources include, lights, power equipment and people in the work area.

Methods used to calculate these heat gains will be discussed in this section. Power requirements and diversity factors as estimated by Black & Veatch (8) will be used.

a. Lights

Lights in the work area are a source of heat gain. This heat gain will be calculated by using the equation below (31) and the fact that 1 watt is equal to 3.41 BTUH (23). (heat gain) = (light - watts) x (diversity factor) x (3.41 BTUH/watt).

b. Equipment

Equipment in the work area give off heat during operation. This heat is also dependent on the power consumed in watts. (heat gain) = (power - watts) x (diversity factor) x (3.41 BTUH/watt).

c. People

Workèrs in the plant give off heat both sensible and latent. For sedentary workers, the values below were found. (5)

Sensible Heat Gain = 275 BTU/Hr/Person

Latent Heat Gain = 525 BTU/Hr/Person

For people doing slightly more vigorous work, the values below were found. (5)

Sensible Heat Gain = 305 BTU/Hr/Person

Latent Heat Gain = 595 BTU/Hr/Person

These values will be used where appropriate to determine heat gains.

Infiltration

The areas under consideration are operated at negative pressure so that dust laden air will not infiltrate other work areas. Infiltration of outside air into these areas through doorways will result because of this. Heat gain will result due to the higher temperature and humidity content of the outside air. This heat gain can be broken down into sensible and latent gains as shown below. (5)

Sensible $q_s$ = Infiltration (cfm) x 1.08 ( T)

T = T outside - T inside

Latent $q_l$ = Infiltration (cfm) x 4840 ( W )

W = W outside - W inside

Where T = Temperature

W = Moisture content (lbw/lba)

Tables 3-1 through 3-4 will show the total heat gains for the areas under study.

## PRIMER INSERT

### Effective Transmission Gains

| | Source | U | Area ($ft^2$) | T | BTUH |
|---|---|---|---|---|---|
| (factory) | wall | .28 | 720 | 28 | 5,645 |
| (factory) | wall | .32 | 1200 | 28 | 10,752 |
| (factory) | wall | .15 | 648 | 15 | 1,458 |
| | roof | .18 | 6300 | 28 | 31,752 |
| (outside) | doors | .40 | 154 | 15 | 924 |
| (factory) | doors | .40 | 175 | 28 | 1,960 |

### Internal Gains

| | Source | | | BTUH |
|---|---|---|---|---|
| (sensible) | People | 20 people x 275 BTU/person | = | 5,500 |
| (latent) | People | 20 people x 525 BTU/person | = | 10,500 |
| | Lights | 47,250w x .9 x 3.41 | = | 145,010 |
| | Equipment | 245,000w x .8 x 3.41 | = | 668,360 |

### Infiltration

BTUH

Infiltration = 1 air exchange per hour

1 x 75,600 $ft^3/hr$ = 1,260 cfm

(sensible) $q_s$ = 1,260 cfm x 1.08 ($101^\circ - 73^\circ$) = 38,102

(latent) $q_l$ = 1,260 cfm x 4,840 (.01542 - .00865)= 41,286

### Totals

$q_s$ = 909,463 BTUH
$q_l$ = 51,786 BTUH
$q_t$ = 961,249 BTUH

$\frac{q_s}{q_t}$ = .9461

Table 3-1

# LOAD & ASSEMBLY

## Effective Transmission Gains

| | Source | U | Area ($ft^2$) | T | BTUH |
|---|---|---|---|---|---|
| (outside) | wall | .15 | 1056 | 15 | 2,376 |
| (factory) | wall | .32 | 1260 | 28 | 11,290 |
| (factory) | roof | .17 | 5495 | 28 | 26,156 |
| (outside) | doors | .40 | 42 | 15 | 252 |
| (factory) | doors | .40 | 213 | 28 | 2,386 |

## Internal Gains

| | Source | | | BTUH |
|---|---|---|---|---|
| (sensible) | People | 20 people x 275 BTU/person | = | 5,500 |
| (latent) | People | 20 people x 525 BTU/person | = | 10,500 |
| | Lights | 41,312w x .9 x 3.41 | = | 126,481 |
| | Equipment | 275,000w x .9 x 3.41 | = | 750,200 |

## Infiltration

Infiltration = 1 air exchange per hour

$1 \times 65{,}940\ ft^3/hr = 1{,}099$ cfm

| | | BTUH |
|---|---|---|
| (sensible) | $q_s = 1{,}099 \text{ cfm} \times 1.08\ (101^\circ - 73^\circ) =$ | 33,233 |
| (latent) | $q_l = 1{,}099 \text{ cfm} \times 4{,}840\ (.01542 - .00865) =$ | 36,010 |

## Totals

$q_s = 957{,}874$ BTUH

$q_l = 46{,}510$ BTUH

$q_t = 1{,}004{,}384$ BTUH

$$\frac{q_s}{q_t} = .9537$$

Table 3-2

# TRACER CHARGE

## Effective Transmission Gains

| | Source | U | Area ($ft^2$) | T | BTUH |
|---|---|---|---|---|---|
| (outside) | wall | .15 | 300 | 15 | 675 |
| (factory) | wall | .32 | 1096 | 28 | 9,820 |
| (factory) | roof | .17 | 823 | 28 | 3,918 |
| (factory) | doors | .40 | 80 | 28 | 896 |

## Internal Gains

| | Source | | | BTUH |
|---|---|---|---|---|
| (sensible) | People | 11 people x 305 BTU/person | = | 3,355 |
| (latent) | People | 11 people x 645 BTU/person | = | 7,095 |
| | Lights | 53,495w x .9 x 3.41 | = | 16,418 |
| | Equipment | 80,000w x .8 x 3.41 | = | 218,240 |

## Infiltration

• Infiltration = $\frac{1}{2}$ air exchange per hour

$\frac{1}{2}$ x 4,938 $ft^3/hr$ = 82.3 cfm

| | | BTUH |
|---|---|---|
| (sensible) | $q_s$ = 82.3 cfm x 1.08 ($101^\circ - 73^\circ$) = | 2,489 |
| (latent) | $q_l$ = 82.3 cfm x 4,840 (.01542 - .00865) = | 2,697 |

## Totals

$q_s$ = 255,811 BTUH
$q_l$ = 9,792 BTUH
$q_t$ = 265,603 BTUH

$$\frac{q_s}{q_t} = .9631$$

Table 3-3

## SAMPLE BULLET ASSEMBLY

### Effective Transmission Gains

| | Source | U | Area ($ft^2$) | T | BTUH |
|---|---|---|---|---|---|
| (outside) | wall | .15 | 405 | 15 | 912 |
| (factory) | wall | .32 | 1558 | 28 | 13,960 |
| (factory) | roof | .17 | 600 | 28 | 2,856 |
| (outside) | roof | .05 | 1000 | 63 | 3,150 |
| (outside) | doors | .40 | 35 | 15 | 210 |
| (factory) | doors | .40 | 202 | 28 | 2,262 |

### Internal Gains

| | Source | | | BTUH |
|---|---|---|---|---|
| (sensible) | People | 3 people x 305 BTU/person | = | 915 |
| (latent) | People | 3 people x 645 BTU/person | = | 1,935 |
| | Lights | 10,400w x .9 x 3.41 | = | 31,918 |
| | Equipment | 10,000w x 1 x 3.41 | = | 34,100 |

### Infiltration

Infiltration = 1.5 air exchange per hour

1.5 x 19,200 $ft^3/hr$ = 480 cfm

| | | BTUH |
|---|---|---|
| (sensible) | $q_s = 480 \text{ cfm} \times 1.08 \; (101^\circ - 73^\circ) =$ | 14,515 |
| (latent) | $q_l = 480 \text{ cfm} \times 4,840 \; (,01542 - .00865) =$ | 15,728 |

### Totals

$q_s$ = 104,788 BTUH
$q_l$ = 17,663 BTUH
$q_t$ = 122,451 BTUH

$$\frac{q_s}{q_t} = .8558$$

Table 3-4

## CHAPTER IV

## COOLING REQUIREMENTS

Air conditioning equipment costs are a significant portion of the total building construction cost, running as high as 10% of the total cost in office buildings and schools. (14) In industrial applications, where contaminant control dictates high air exchange rates, this cost can be significantly higher. Care should be taken in selecting the air conditioning system design so that this portion of the cost can be kept as low as possible without sacrificing worker safety. Air conditioning system designs using various percentages of recirculation will be considered in Chapter 4. Required refrigeration and air cleaner capacity will be determined for each of these designs.

### Required Air Exchange Rates

It is recommended that the volume flow rate of exchange air be determined first in air conditioning calculations. The required temperature of the return air and the amount of cooling can then be found using these values. (31)

Equation 3-2 is used to determine the required air exchange rate for various degrees of recirculation. Certain modifications will be needed to make it applicable. These modifications will be shown below.

In a system using recirculation, some air, with entrained dust, that is exhausted from the room will be returned as ventilation air. The amount of dust that is returned depends upon the air exchange rate, the dust concentration, filter efficiency and the degree of recirculation. This returned dust can be considered an additional source of dust emission. When this emission is added to the original generation rate, a new effective generation rate is determined. This new rate is given below.

$$G'' = G' + RQC(1-\eta) \qquad 4\text{-}1$$

Where:
$G''$ = Total Generation
$G'$ = Original Generation
$R$ = Percent Recirculation
$Q$ = Air Exchange Rate
$C$ = Dust Concentration
$\eta$ = Air Washer Efficiency

The dust removal rate will remain the same for this case. Using the fact that the rate of generation is equal to the rate of dust removal for constant workroom dust concentrations, the relation below is obtained.

$$G' + RQC(1-\eta) = QC \qquad 4\text{-}2$$

This equation can be solved for Q as below.

$$Q = \frac{G'}{C(1-R+R\eta)} \qquad 4\text{-}3$$

Equation 4-4 will be used to determine the air exchange rates needed to provide a dust concentration of .03mg/$M^3$ in the areas of consideration for various percentages of recirculation. Values of C, $G'$ and $\eta$ will be taken from Chapter 3 and used to obtain the results in Table 4-1.

It should be noted that the value of $G'$ in this equation is not the actual generation rate of the processes, as this value was not obtainable. This value is the maximum acceptable generation rate determined in Chapter 3. It is conservatively chosen so that it is always larger than the actual value.

Return Air Conditions

Return air must be conditioned before re-entering the room so that it is able to compensate for the heat and moisture gains of the area under consideration. The dry-bulb temperature of the return air for a given air flow rate can be determined by using the equation below from Threlkeld. (31)

$$q_s = m_a \, c_{pa} \, (t_2 - t_1)$$

Where: $q_s$ = Sensible heat gain

$m_a$ = Mass of dry air returned/hr.

$c_{pa}$ = Specific heat for dry air

= .245 BTU/lba - $^{o}F$

$t_2$ = Room design D.B. temperature

$t_1$ = Return air D.B. temperature

REQUIRED AIR EXCHANGE RATES

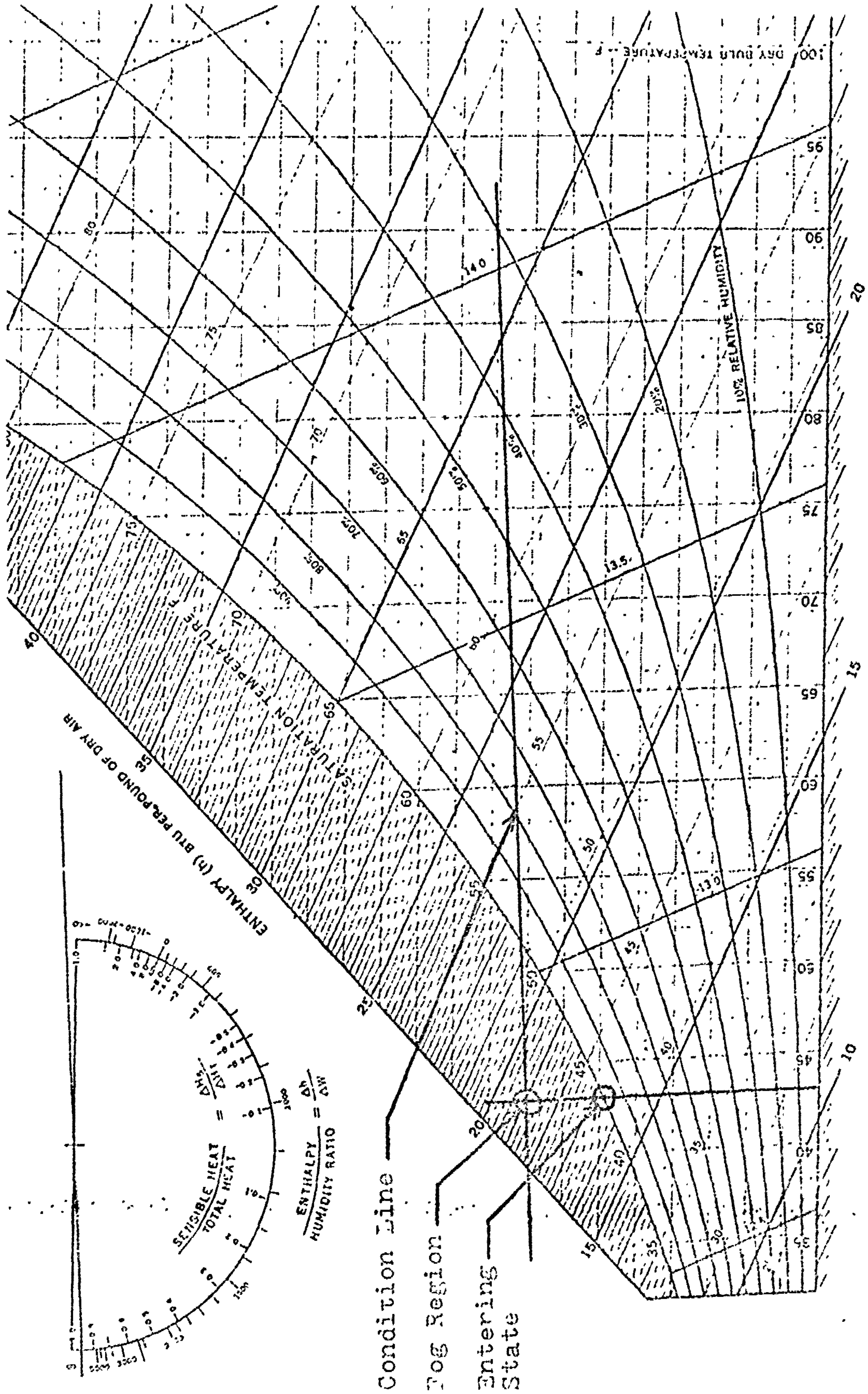
| Recirculation % | Primer Insert cfm | Load & Assembly cfm | Tracer Charge & Sample Bullet cfm | Total cfm |
|---|---|---|---|---|
| 0 | 32,101 | 30,000 | 10,999 | 73,100 |
| 10 | 32,327 | 30,211 | 11,076 | 73,614 |
| 20 | 32,557 | 30,426 | 11,155 | 74,138 |
| 25 | 32,673 | 30,534 | 11,195 | 74,402 |
| 30 | 32,789 | 30,644 | 11,235 | 74,668 |
| 40 | 33,026 | 30,864 | 11,316 | 75,206 |
| 50 | 33,265 | 31,088 | 11,398 | 75,751 |
| 60 | 33,508 | 31,315 | 11,481 | 76,304 |
| 70 | 33,755 | 31,546 | 11,566 | 76,867 |
| 75 | 33,880 | 31,662 | 11,608 | 77,150 |
| 80 | 34,005 | 31,780 | 11,651 | 77,436 |
| 90 | 34,260 | 32,017 | 11,738 | 78,015 |
| 100 | 34,517 | 32,258 | 11,827 | 78,602 |

Table 4-1

The values of $q_s$ are the sensible heat gains of the areas determined in Chapter 3. They are increased by 10% to make up for heat gain of the air due to work input by the fans or air moving equipment. Required return air temperatures are shown in Table 4-2.

The above equation gives only the required dry-bulb temperature of the return air so that sensible heat gain is compensated. The moisture content of the return air must also be determined so that the latent heat gain of the room will be balanced. Threlkeld suggests the use of sensible heat ratio (S.H.R.) to determine the total return air conditions. (31) This S.H.R. is equal to $\frac{q_s}{q_s + q_l}$ and is used to determine the slope of the condition line shown in Figure 4-1. This slope is taken from the protractor in the upper left corner of Figure 4-1. (31) A line with this slope is drawn through the point on the psychrometric chart representing the room design conditions as shown in Figure 4-1. (31) This determined condition line represents all combinations of return air conditions that can be used to exactly balance the heat and moisture gains of the desired area. The state of the return air can be determined by finding the intersection of the required dry-bulb temperature line and the condition line as shown in Figure 4-1.

In all cases, this intersection was found to be in the fog region (31) as shown in Figure 4-1. Return air is not able to return in this state for this air would contain



ENTHALPY (h) BTU PER POUND OF DRY AIR
SATURATION TEMPERATURE °F
DRY BULB TEMPERATURE °F
10% RELATIVE HUMIDITY
SENSIBLE HEAT / TOTAL HEAT
ENTHALPY / HUMIDITY RATIO
Condition Line
Fog Region
Entering State


Figure 4-1

## AIR CONDITIONS LEAVING COOLER

| Recirculation | Primer Insert | | Load & Assembly | | Tracer Charge & Sample Bullet | |
|---|---|---|---|---|---|---|
| % | T | h | T | h | T | h |
| 0 | 44.10 | 17.30 | 40.44 | 15.50 | 39.56 | 15.05 |
| 10 | 44.31 | 17.40 | 40.66 | 15.61 | 39.80 | 15.16 |
| 20 | 44.51 | 17.51 | 40.89 | 15.71 | 40.03 | 15.28 |
| 25 | 44.61 | 17.56 | 41.01 | 15.77 | 40.15 | 15.33 |
| 30 | 44.71 | 17.62 | 41.12 | 15.82 | 40.26 | 15.39 |
| 40 | 44.91 | 17.72 | 41.35 | 15.93 | 40.50 | 15.50 |
| 50 | 45.12 | 17.82 | 41.58 | 16.04 | 40.73 | 15.61 |
| 60 | 45.32 | 17.93 | 41.80 | 16.14 | 40.96 | 15.72 |
| 70 | 45.52 | 18.04 | 42.03 | 16.25 | 41.20 | 15.84 |
| 75 | 45.62 | 18.09 | 42.14 | 16.30 | 41.32 | 15.90 |
| 80 | 45.72 | 18.14 | 42.26 | 16.36 | 41.43 | 15.95 |
| 90 | 45.92 | 18.24 | 42.49 | 16.46 | 41.67 | 16.06 |
| 100 | 46.13 | 18.35 | 42.71 | 16.57 | 41.90 | 16.17 |

T = $^{o}$F          h = BTU/lba

Table 4-2

condensed water droplets. It will thus, enter the room as saturated air corresponding to the given dry-bulb temperature. The enthalpy of this air is shown in Table 4-2.

As indicated by Figure 4-1, the return air contains less moisture than required to maintain the room relative humidity of 50%. Humidification equipment must be used to add moisture to the air so that the humidity level is maintained. Humidification equipment, needed to supplement winter heating, will be assumed amply large to handle the humidification requirements for all systems under consideration.

Air Mixture Conditions

In order to calculate the refrigeration requirements, the conditions of the air entering the cooling unit must be known. When recirculation is utilized, these conditions depend upon the make-up and recirculated air conditions along with the rate of recirculation.

Heat recovery units are often used to allow the exhaust air to cool make-up air before entering the cooling unit. When these units are used, they will be assumed to have an efficiency of 60%. (8) Air mixture conditions will be calculated for systems with and without such recovery units. The equations below, from Threlkeld, (31) will be used to calculate the air conditions after mixing.

$$ma_1h_1 + ma_2h_2 = ma_3h_3 \qquad 4\text{-}5$$

$$ma_1W_1 + ma_2W_2 = ma_3W_3 \qquad 4\text{-}6$$

Where: $m_w$ = Mass water

$m_a$ = Mass of dry air - lba

h = Enthalpy of air BTU/lba

w = Humidity ratio $m_w/m_a$

1 = Make-up air

2 = Recirculated air

3 = Mixture

These equations can be used only after the conditions of the make-up and recirculation air are known. Conditions of the make-up air are quite readily determined. For systems without heat recovery, the conditions are D.B. = 101 °F and W.B. = 78 °F. Systems using heat recovery have a D.B. of 84.2 °F and a W.B. of 73.4 °F. Enthalpy and humidity ratios can be determined for these states by using psychrometric charts.

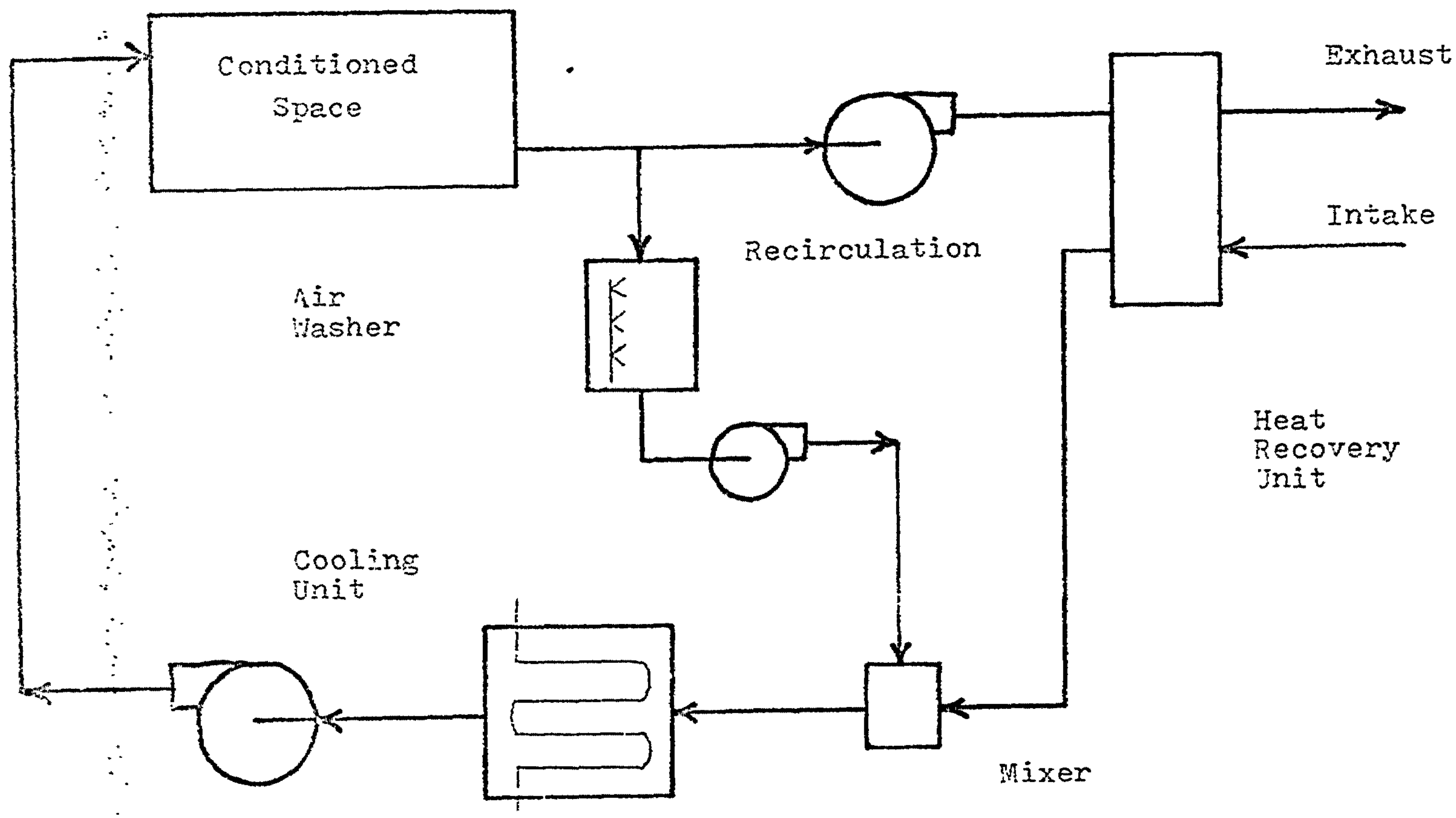
Conditions of recirculated air require more explanation. Recirculated air leaves the design space with design conditions of 73° D.B. and a relative humidity of 50%. This air then enters the air cleaning system where it contacts large amounts of water. During this cleaning process, the air becomes saturated and cooler due to evaporative cooling. (10) No heat is added in the washer and thus the process is considered adiabatic. Under these conditions, air will leave the washer saturated at 61 $^{o}F$ with a humidity ratio of .0114 lbw/lba.

Cleaning systems of this type require relatively large air moving devices to force air through them. (21) These air moving devices as shown in Figure 4-2 are located after the cleaning units and as such will impart heat to the air after leaving the washer. This added heat will be assumed to be sufficient to raise the recirculated air temperature to 73$^{o}$. In reality, the heat gain will not raise the air temperature this much but since it is undesirable to rely upon the air washer for any portion of air cooling, the temperature of 73$^{o}$ will be used. Enthalpy can be determined for this air from psychrometric charts as before.

The mass of recirculation and make-up air can now be determined for the various systems under consideration. The recirculation percentages stated earlier are in terms of air volumes and thus corrections will have to be made to obtain the percentages in terms of air mass. Once this is done, the enthalpy and humidity ratios can be determined for the air mixtures by using equations 4-5 and 4-6 respectively. These values are shown in Table 4-3.

Refrigeration Requirements

The refrigeration requirements can be calculated once the conditions of air entering the cooling coils and the required conditions of the leaving air are determined, as previously shown. By using these enthalpies and total



Conditioned Space
Air Washer
Recirculation
Exhaust
Intake
Heat Recovery Unit
Cooling Unit
Mixer


Figure 4-2

## AIR MIXTURE CONDITIONS ENTERING COOLER

| Recirculation % | With Heat Recovery h BTU/lba | With Heat Recovery W lbw/lba | Without Heat Recovery h BTU/lba | Without Heat Recovery W lbw/lba |
|---|---|---|---|---|
| 0 | 37.10 | .0154 | 41.30 | .0154 |
| 10 | 36.37 | .0149 | 40.11 | .0150 |
| 20 | 35.65 | .0146 | 38.93 | .0146 |
| 25 | 35.29 | .0144 | 38.35 | .0144 |
| 30 | 34.93 | .0142 | 37.77 | .0142 |
| 40 | 34.21 | .0138 | 36.62 | .0137 |
| 50 | 33.50 | .0134 | 35.49 | .0133 |
| 60 | 32.79 | .0130 | 34.36 | .0129 |
| 70 | 32.09 | .0126 | 33.26 | .0126 |
| 75 | 31.74 | .0124 | 32.70 | .0124 |
| 80 | 31.39 | .0122 | 32.16 | .0122 |
| 90 | 30.69 | .0118 | 31.07 | .0118 |
| 100 | 30.00 | .0114 | 30.00 | .0114 |

Table 4-3

required air mass flow rates, the required refrigeration can be calculated. The equation below taken from Threlkeld (31) is used in these calculations.

$$q = \frac{m_a\ (h_2 - h_1)}{12{,}000} \text{ in Tons}$$

Where: $q$ = Tons refrigeration

$m_a$ = Mass flow rate (lba/hr.)

$h_2$ = Enthalpy of entering air (BTU/lba)

$h_1$ = Enthalpy of leaving air (BTU.lba)

12,000 = No. of (BTU/hr.) per ton

Table 4-4 shows the required refrigeration for the given areas. Table 4-5 shows the total refrigeration and air cleaning needs both with and without heat recovery.

REQUIRED REFRIGERATION

(With Heat Recovery)

| Recirculation % | Primer Insert Tons | Load & Assembly Tons | Tracer Charge & Sample Bullet Tons |
|---|---|---|---|
| 0 | 233.2 | 237.7 | 88.9 |
| 10 | 225.0 | 230.1 | 86.2 |
| 20 | 216.6 | 222.6 | 83.4 |
| 25 | 212.5 | 218.6 | 81.9 |
| 30 | 208.2 | 214.8 | 80.5 |
| 40 | 199.8 | 207.0 | 77.7 |
| 50 | 191.3 | 199.1 | 74.8 |
| 60 | 182.7 | 191.3 | 71.9 |
| 70 | 174.0 | 183.3 | 68.9 |
| 75 | 169.6 | 179.3 | 67.5 |
| 80 | 164.8 | 175.2 | 66.0 |
| 90 | 156.5 | 167.1 | 63.0 |
| 100 | 147.5 | 158.9 | 60.0 |

Table 4-4

TOTAL AIR CONDITIONING REQUIREMENTS

| Recirculation % | Refrigeration With Recovery Tons | Refrigeration Without Recovery Tons | Air Cleaner cfm |
|---|---|---|---|
| 0 | 559.8 | 672.4 | 73,100 |
| 10 | 541.3 | 642.3 | 73,614 |
| 20 | 522.6 | 611.8 | 74,138 |
| 25 | 513.0 | 596.5 | 74,402 |
| 30 | 503.5 | 581.3 | 74,668 |
| 40 | 484.5 | 550.9 | 75,206 |
| 50 | 465.2 | 520.5 | 75,751 |
| 60 | 445.9 | 489.8 | 76,304 |
| 70 | 426.2 | 459.2 | 76,867 |
| 75 | 416.4 | 443.7 | 77,150 |
| 80 | 406.0 | 428.4 | 77,436 |
| 90 | 386.6 | 397.5 | 78,015 |
| 100 | 366.4 | 366.3 | 78,602 |

Table 4-5

# CHAPTER V

## COST ANALYSIS

From the results obtained in Chapter 4, it can be seen that the required refrigeration decreases and the required air washer capacity increases with increasing percentages of recirculation. Although the rate of refrigeration reduction is more significant than the rate of air washer capacity increase, a decision as to which system design is best cannot be made from this data. Total system costs will b determined so a comparison and hence, decision can be made.

Although three cooling coils are required in this facility, each will be supplied chilled water from one central cooling unit. Refrigeration costs will be based upon the total refrigeration load as seen by the central cooling unit. Three air washer systems will also be required. This cost will be the sum of the three individual cleaner costs as determined for each system design.

Modern Air Conditioning Heating & Ventilating, (10) listed average installed air conditioning equipment costs for industrial applications. These costs included all necessary equipment such as, heat exchangers, chillers, pumps, fans, piping and regular air filters. Although the actual costs could vary by as much as 30% on either side of this average value, (10) the indicated average values were used in this study. Cost indexes in "Engineer News Record" (14) were used to correct the cost for 1975 prices. It was determined that installed air conditioning cost was $1781 per ton of refrigeration.

This figure was found to be typical of air conditioning costs for various facilities listed in Engineer News Record. (15) Costs, of such equipment, however, ranged from $900 per ton for an office building to $2300 per ton for some industrial facilities. (15)

Cost estimates for wet air cleaners of the type used in this facility were obtained from Jones. (21) The same cost indexes mentioned above were used to correct the costs for 1975 prices. Air cleaning costs for each area are indicated in Table 5-1. Table 5-2 is a list of the total estimated cost of air conditioning equipment. Operating cost estimates were also determined by using estimated pump, fan, and compressor power requirements as shown in Modern Air Conditioning Heating & Ventilating. (10) Results are shown in Table 5-2.

It is seen that the 75% recirculation design is a feasible alternative to the 100% make-up unit from an economic point of view. The total installed equipment cost is found to be almost $230,000 less than the original design. This represents a significant savings while sacrificing no system performance. Although this design is preferred to the 100% make-up air unit, it is not the optimal design. The design using 100% recirculation is the least expensive.

Because outside air must be brought in to replenish

AIR CLEANING EQUIPMENT COSTS

| Recirculation % | Primer Insert $ | Load & Assembly $ | Tracer Charge & Sample Bullet $ | Total $ |
|---|---|---|---|---|
| 0 | 115,162 | 108,032 | 55,919 | 279,113 |
| 10 | 115,929 | 108,748 | 56,147 | 280,824 |
| 20 | 116,710 | 109,478 | 56,381 | 282,569 |
| 25 | 117,103 | 109,844 | 56,500 | 283,447 |
| 30 | 117,497 | 110,217 | 56,619 | 284,333 |
| 40 | 118,301 | 110,964 | 56,859 | 286,124 |
| 50 | 119,112 | 111,724 | 57,102 | 287,938 |
| 60 | 119,937 | 112,495 | 57,348 | 289,780 |
| 70 | 120,775 | 113,279 | 57,600 | 291,654 |
| 75 | 121,200 | 113,672 | 57,725 | 292,597 |
| 80 | 121,623 | 114,072 | 57,853 | 293,548 |
| 90 | 122,489 | 114,877 | 58,111 | 295,477 |
| 100 | 123,361 | 115,695 | 58,375 | 297,431 |

Table 5-1

TOTAL SYSTEM COSTS

| Recirculation % | Refrigeration $ | Total $ | Operating $ |
|---|---|---|---|
| 0 | 997,004 | 1,276,117 | 14.68 |
| 10 | 964,055 | 1,244,879 | 14.25 |
| 20 | 930,751 | 1,213,320 | 13.86 |
| 25 | 913,653 | 1,197,100 | 13.63 |
| 30 | 896,734 | 1,181,067 | 13.37 |
| 40 | 862,894 | 1,149,018 | 12.91 |
| 50 | 828,521 | 1,116,459 | 12.44 |
| 60 | 794,148 | 1,083,928 | 11.97 |
| 70 | 759,062 | 1,050,716 | 11.48 |
| 75 | 741,608 | 1,034,205 | 11.23 |
| 80 | 723,086 | 1,016,634 | 10.97 |
| 90 | 688,535 | 984,012 | 10.45 |
| 100 | 652,558 | 949,989 | 9.92 |

Table 5-2

oxygen and remove carbon dioxide, the 100% recirculation design cannot be used. (10) Outside air requirements are quite small, being about 50 cfm per person. (31) The system design utilizing 90% recirculation will be more than adequate in this respect and will be assumed to be the optimal acceptable design. A savings of $280,000 is realized by using this system instead of the original 100% make-up design.

Several important observations will be discussed to conclude this study. As stated earlier, air flow rates that would provide equivalent dust concentrations were determined. The air flow rates in this study, did not increase significantly as the recirculation rate increased because a high efficiency air washer was chosen. If a lower efficiency washer was used, an important difference would occur. The air flow rate would have to be increased significantly to meet the design concentration. This would result in a larger cooling load and would require a larger capacity air washer. The combination of these effects could possibly change the optimum system. The air washer should be selected with an efficiency as high as possible when using recirculation.

In the facility under study, both the recirculated air and exhaust air had to be cleaned. As a result of this fact, only a small increase in the required air cleaner capacity was necessitated by an increase in the recirculation rate. If the system did not require exhaust air cleaning, this capacity would change significantly. Requirements would be zero in a 100% make-up air design and 85,000 cfm in a 100% recirculation design. The result of this would be similar system costs for 0 and 100% recirculation.

It is important to consider all the factors before making a final decision as to which unit to use. It has been shown that acceptable dust concentration could be maintained with systems using recirculation. Dust concentration, however, will increase significantly if the air washer malfunctions when recirculation is being used. Mutchler (22) suggests that recirculation should not be used when dealing with extremely explosive or toxic substances for this reason. The designer or industrial hygienist will have to make the decision as to whether the savings justifies this added risk.

# CHAPTER VI

# CONCLUSION

A procedure has been presented that can be used to compare ventilation systems with various make-up air percentages. Certain restrictions upon the use of such a method, however, will be stated. The given procedure is best suited to systems emitting gaseous or vapor contaminants. The reason for this is that contaminants of this type diffuse in the classical sense and result in a uniform room concentration. Contaminant emission should be of a constant uniform nature. In the application of this method, the contaminant was airborne explosive dust. This dust was emitted at a constant rate and was of a size easily transported by air currents. It was assumed that ventilation duct design was adequate to provide thorough dispersion of this airborne dust. The presented procedure was considered appropriate for these reasons. Care should be taken in deciding whether this method can be used for any given situation.

Selection of design conditions as presented in Chapter 3 is the most important part of such a study. These values will be used throughout the study and are the basis of all calculations. Incorrect selection here, could result in a system far from the most economic. The main steps of such a study will be summarized due to their importance.

The problem area should first be defined including the source and type of contaminant emission. Local ventilation should be considered as a means of controlling this emission. If this is not a practical solution, general ventilation can be used, as in this application. A conservative estimate of the contaminant emission rate should then be determined. In some situations this estimate can be obtained from similar applications. Where previous data is unavailable, engineering skill and theoretical considerations should be used to set a conservative value.

In this study, a dust generation rate was determined using data supplied by Black & Veatch. (8) The calculated generation rate was larger than would actually occur and was considered the maximum acceptable rate. When in operation, workroom air should be monitored to insure that this generation rate is not exceeded. If exceeded, the ventilation system will not be able to maintain desired concentrations and emitting processes will need further control.

Once the type, size and amount of dust emission is known, available air cleaning equipment should be investigated. A unit with high efficiency and safe operation should be chosen. A minimum effective efficiency should be determined for use in the remainder of the study. Appropriate inside and outside environment conditions must be decided upon and internal heat gains determined.

With these values set, required ventilation rates can be calculated for each of the system designs. Required air cleaning capacity and refrigeration can then be determined by methods shown in Chapter 4 & 5. Equipment costs can be estimated and used with these values to determine the total system costs.

The stated procedure provides a comparison of total system costs for designs using various rates of recirculation. Although these costs were used as the sole criteria for deciding upon an optimal design in this study, other considerations must be made. As stated in Chapter 5, consequences of air washer malfunction must be considered for designs using recirculation. The need for safe guarding equipment and monitoring devices should be determined for each application on an individual basis. Costs of such required equipment should be included in the total system cost. Finally, the monetary savings should be weighed against added risk to see if recirculation is justified. The decision as to which system is optimal for the given application can be made only after all these factors are considered.

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